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AFIT/GA/AA/88D-14

AN EXPERIMENTAL METHOD OF TESTING THIN-FILM HEAT TRANSFER GAGES

THESIS

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AN EXPERIMENTAL METHOD OF TESTING THIN-FILM HEAT TRANSFER GAGES

THESIS

Presented to the Faculty of the School of Engineering

of the Air Force Institute of Technology

Air University

In Partial Fulfillment of the

Requirements for the Degree of

Master of Science in Astronautical Engineering

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December 1988

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Preface

The purpose of this study is to assist those who use thin-film gages for the purpose of determining heat transfer rates and heat transfer coefficients. On occasion, the gages have been used for relatively long times (up to 10 seconds) when gage accuracy for those long times was unknown. Hopefully, this study will help users understand the problems involved and provide guidelines for obtaining accurate results from the gages.

Without the valuable help of many people, this thesis would never have been completed. Time after time, my advisor, Dr. James E. Hitchcock, steered me through the maze of experimental pitfalls, several incidents of which are described in detail in this thesis. His patience and stability were a constant encouragement to me. Dr. William C. Elrod was a great help in arranging laboratory facilities for the experiment. Thanks also go out to two expert laboratory technicians. Jay Anderson set up the instrumentation and educated me in its use, while John Brohas fabricated the nozzle from aluminum stock in only two days.

Finally, I want to thank my wife, Doreen, for her patience and faithful support throughout this eighteen month program.

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English Symbols

A - Area, in^2

c_p - Specific heat at constant pressure, Btu/lb_m-^oR

E - Voltage, Volt

G - Amplifier gain

g_c - Newton's Second Law constant, 32.2 lb_m-ft/lb_f-s^2

h - Heat transfer coefficient, Btu/ft^2-sec-^oR

I - Electrical current, Ampere

k - Thermal conductivity, Btu/ft-sec-^oR

P - Power, Btu/hr

Pr - Prandtl number

q - Heat flux, Btu/ft^2-sec

R - Electrical resistance, Ohm

Re_x - Reynolds number based on distance along the surface

$Re_{\Delta 2}$ - Reynolds number based on enthalpy thickness

r - Radius of nozzle, in

St - Stanton number

T_{AW} - Adiabatic wall temperature, ^oR

T_e - Temperature at the edge of the boundary layer, ^oR

T_s - Surface temperature, ^oR

T^* - Eckert's reference temperature, ^oR

t - Time, sec

u_e - Velocity at the edge of the boundary layer, ft/sec x - Distance measured along the surface, in

<u>Greek Symbols</u>

α - Thermal diffusivity, ft^2/sec

δ - Thermal disturbance distance, in

Δ_2 - Enthalpy thickness, in

μ - Dynamic viscosity, $lb_f\text{-}sec/ft^2$

ν - Kinematic viscosity, ft^2/sec

ρ_e - Air density at boundary layer edge, lb_m/ft^3

Abstract

Thin-film heat transfer gages, consisting primarily of a vacuum-deposited platinum film on a quartz substrate, have been used successfully for measuring the rapid transients in shock tubes with a time duration of a few milliseconds. For these short times, a thermal disturbance propagates only a small distance from the surface of the gage, permitting a one-dimensional, semi-infinite solid analysis to be used to convert surface temperatures into surface heat fluxes. However, results can be inaccurate due to two primary factors: (1) the gage is preheated by the current necessary for operation, and (2) extended operating times allow the thermal disturbance to travel deeper into the quartz substrate of the gage, allowing two-dimensional heat transfer and reducing the accuracy of the one-dimensional analysis.

With the use of thin-film gages, heat transfer coefficients were determined experimentally for turbulent choked flow through a nozzle for experimental runs of 0.1, 1.0, and 10.0 seconds. A laminar run of 1.0 second duration was also conducted. These experimental results were compared with the theoretical heat transfer coefficient for each case. The comparisons yielded time duration estimates for which the gages can be considered reliable and gave estimates of the error incurred by relatively long runs. The effects of preheating were examined by increasing the

operating current and again comparing experimental and theoretical heat transfer coefficients.

AN EXPERIMENTAL METHOD OF TESTING THIN-FILM HEAT TRANSFER GAGES

I. Introduction

A primary factor in the development of thin-film heat transfer gages was the need for accurate experimental data in shock tube research [Reference 1]. Instruments before the early 1950s which were ideal for steady-state experiments were unable to respond accurately to the rapidly changing environment in a shock tube. However, in 1956, the "thin-film thermometer", the forerunner of the thin-film heat transfer gage, was introduced. It was extremely sensitive to temperature changes and had a response time of 0.05 microseconds.

As is the case with any tool, the temptation exists to use the gages in applications for which they were not designed. Although the gages have extremely fast response times, appropriate for shock tube experimentation, their reliability in measuring heat transfer in a longer-duration experiment is in question. Lileikis [Reference 2], for example, used these gages for tests of nine seconds' duration.

Also, when measuring small heat transfer rates, there is the temptation to increase the electrical current through the gages so as to increase their natural output signal and reduce noise, thus requiring less amplification.

The objectives of this experimental research are to determine the test time and electrical current limitations of typical thin-film heat transfer gages. To accomplish these objectives, it is proposed to subject the gages to the sudden application of a reasonably constant, known heat flux. The gage limitations should then be readily apparent.

The apparatus chosen for the experiment is a modified ASME convergent nozzle through which ambient air flows to a vacuum tank. The nozzle is plugged, evacuated, and then heated. A test is initiated by suddenly pulling the plug. At the nozzle throat, two thin-film heat gages are mounted. Gage data (surface temperature versus time) is then analyzed using a one-dimensional mathematical model (discussed in detail in Chapter 2) to determine the heat transfer coefficient as a function of time. This one-dimensional characterization will be shown to be the primary source of error in the experimental method.

Theoretically, the heat transfer coefficient remains constant in time, since it is a fluid mechanical property of the system. Given the stagnation conditions and the geometry of the choked nozzle, a constant value for the heat transfer coefficient may be determined theoretically. The integral method of doing this is also discussed in Chapter 2.

The results of these experiments and a discussion of thin-film heat transfer gage limitations are presented in Chapter 4, while a description of the experimental apparatus and procedure is presented in Chapter 3. Computer programs used to calculate both the experimental and theoretical values of the heat transfer rates are included in Appendix B.

II. Theoretical Background

Thin Film Gage Theory

Modern thin-film heat transfer gages consist of a thin film of platinum on a substrate which is usually made of quartz. As the temperature of the platinum film is varied, its electrical resistance changes according to the following relation:

$$R_g(T) = R_g(T_{ref}) + (T - T_{ref})(\Delta R/\Delta T)_g \qquad (2\text{-}1)$$

The quantity $(\Delta R/\Delta T)_g$ is a characteristic of each gage and is found by calibration.

For the experiment of this thesis, the gage was connected in a bridge illustrated by Figure 1. With a constant voltage E_b applied to the bridge, a temperature change on the surface of the gage causes the change of resistance described above. This change of resistance causes a corresponding change in the output voltage of the bridge. During an experimental run, the output voltage of the bridge was measured as a function of time and Equation 2-2 was used to determine the temperature at the surface of the gage as a function of time.

From Reference 1, the change of temperature as a function of change of voltage is given by

$$\Delta T = \frac{[R_2 + R_g(T)]}{(R_2)(\Delta R/\Delta T)_g(I_g)G}\,\Delta E \qquad (2\text{-}2)$$

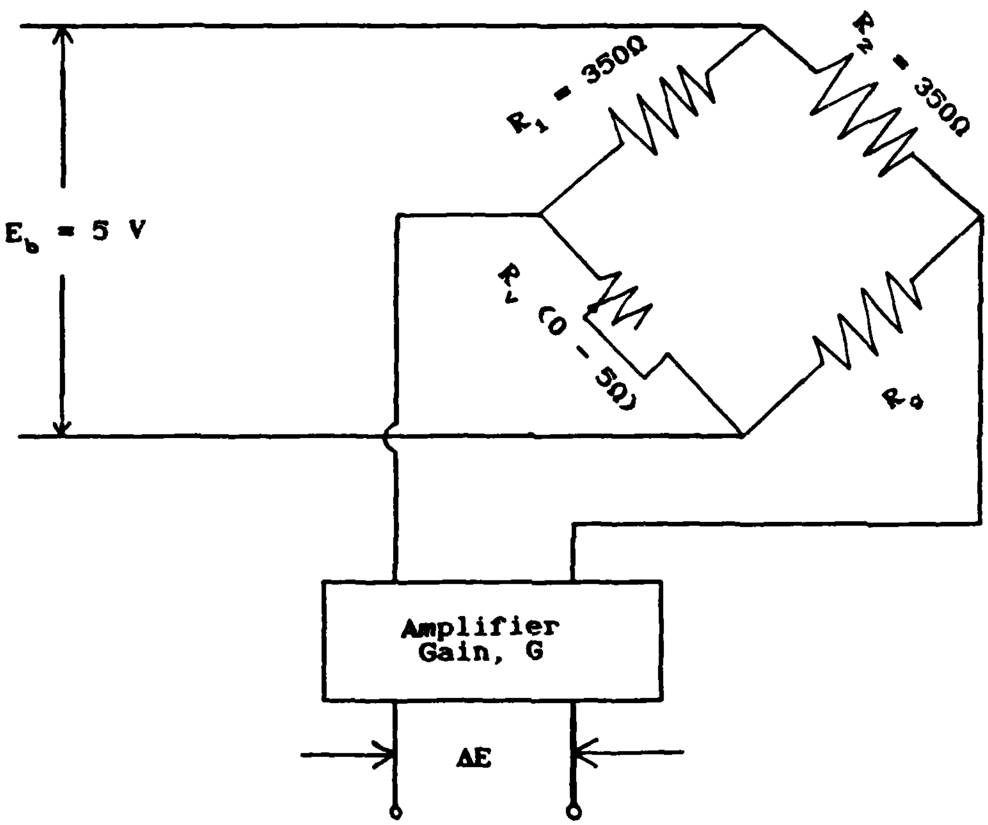

Figure 1: Bridge Circuit Used With Thin-Film Gages

where I_g is the current through the gage.

Experimental Determination of the Heat Transfer Coefficient

A one-dimensional heat conduction equation for a semi-infinite solid is usually used to convert experimental data from thin film gages (surface temperature versus time) to heat transfer rates and heat transfer coefficients as a function of time.

As demonstrated in References 2 and 3, the desired equation for heat flux is

$$q(t_N) = 2\sqrt{\frac{\rho c k}{\pi}}\left[\frac{T_s^+(t_0)}{2(t_N)^{.5}} + \sum_{i=1}^{N} \frac{T_s^+(t_i) - T_s^+(t_{i-1})}{(t_N-t_i)^{.5} + (t_N-t_{i-1})^{.5}}\right] \quad (2\text{-}3)$$

where T_s^+ represents the difference between the surface temperature at the time indicated and the initial temperature. Then, with Newton's law of cooling

$$q(t) = h(t)[T_s(t) - T_{AW}] \quad (2\text{-}4)$$

where

$$T_{AW} = T_e + \frac{r_c u_e^2}{2g_c c_p} \quad (2\text{-}5)$$

$$r_c = Pr^{1/3}\ \text{(turbulent)};\quad r_c = Pr^{1/2}\ \text{(laminar)}$$

one can easily solve for the heat transfer coefficient h as a function of time.

Two problems exist with using the one-dimensional model for data analysis, especially over longer run times. First, the heat flux will not propagate in a one-dimensional fashion as experimental run time increases. As seen in Figure 2, as heat leaves the gage and is absorbed in the air stream over an extended time period, a two-dimensional effect arises as heat flows from the surrounding material into the gage substrate. This occurs because a thermal disturbance travels a distance δ according to

$$\delta \sim \sqrt{\alpha t} \tag{2-6}$$

where α is the thermal diffusivity of a material [Reference 7, page 115-118]. Since the thermal diffusivity for quartz is greater than that for teflon, the thermal disturbance propogates faster through quartz than through teflon.

The two-dimensional effect causes more heat to be available at the surface of the gage (with a correspondingly higher temperature) than the one-dimensional model accounts for. As a result, the heat flux and therefore the heat transfer coefficient as calculated by the one-dimensional model will be lower than the actual values. As the thermal disturbance penetrates deeper into the substrate, this two-dimensional effect becomes more pronounced, and the error in the calculation of the heat transfer coefficient h is increased.

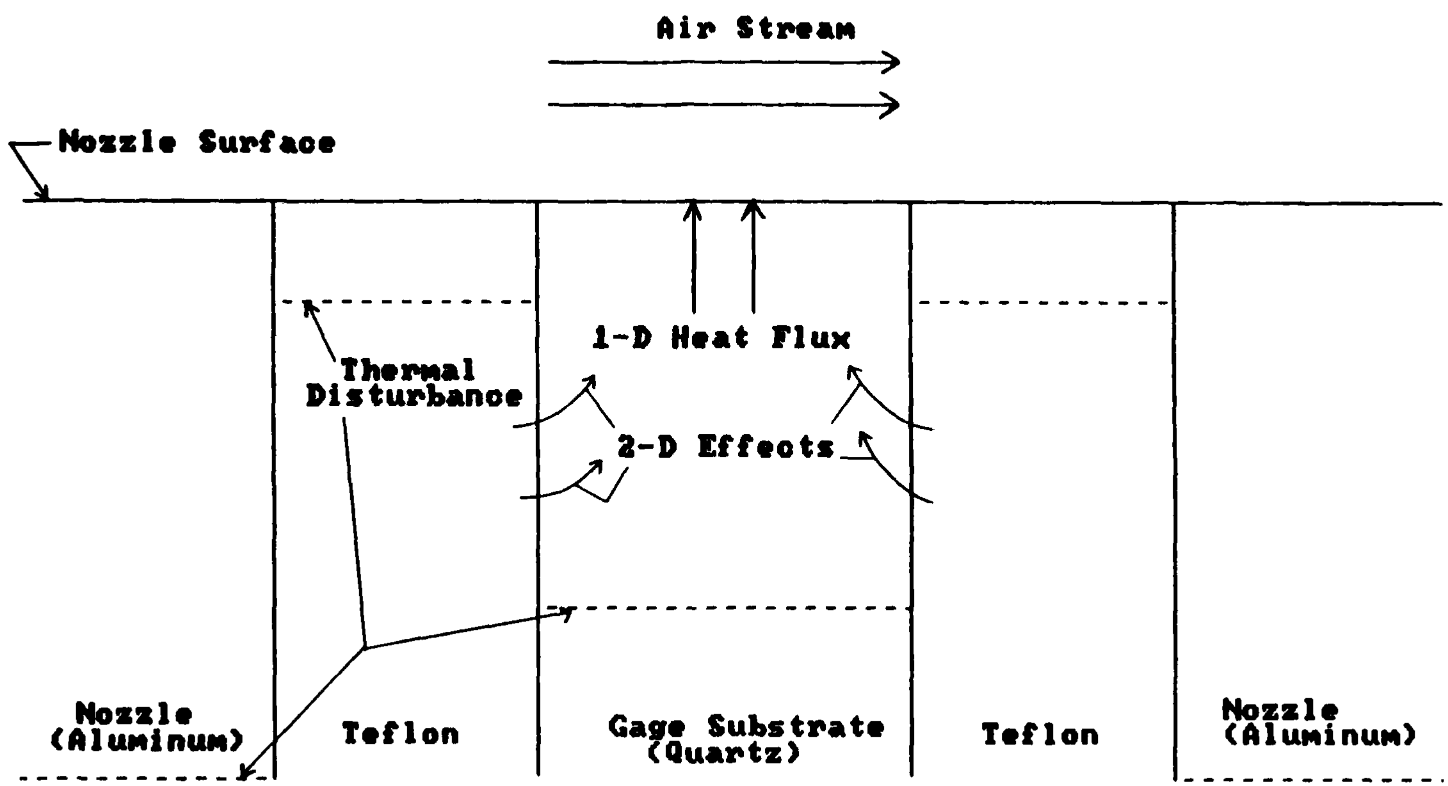

Figure 2: Two-Dimensional Effect in Quartz Substrate

The second problem with the one-dimensional model is that the current through the gage causes the temperature at the surface of the gage to be higher than the temperature of the surrounding nozzle. This effect is called preheating. Kays [Reference 4, page 216] shows that temperature variations along a surface will cause a corresponding change in heat transfer coefficient at those points. If the temperature difference between the surface and the free stream increases, the heat transfer coefficient is increased and vice versa. Therefore, the increased temperature at the gage location should be reflected by a higher heat flux as well as heat transfer coefficient. This is substantiated by the results of Section IV.

Initially, it was believed that temperature gradients in the gage substrate which were caused by the heating effect of the electrical current would cause the preheating effect to change with time. In other words, the effect would initially appear to produce a higher heat transfer coefficient during an experimental run, and then the effect would disappear. However, Bonafede [Reference 6] demonstrated with a numerical analysis that the effect of preheating should be constant throughout a run if the temperature gradient in the substrate does not change with time (i.e., no transients exist). Run #4 of Section IV substantiates Bonafede's findings. More experimental work needs to be done to quantify the relative effects of preheating. Some suggestions are given in Section V.

For comparison with surface heat flux, one can define a heat generation heat flux which is caused by preheating as

$$q_g = P/A = I^2R/A \qquad (2\text{-}7)$$

where A is the surface area of the thin film. In general, heat generation fluxes which are small compared to surface heat flux do not significantly affect the value of the heat transfer coefficient h.

Having described the method of finding h by experiment, a method of calculating h based on theory, to establish a basis of comparison, is described.

Theoretical Determination of the Heat Transfer Coefficient

The Stanton number for turbulent flow with constant surface temperature and constant free-stream velocity over a flat plate as given in Reference 4 may be written as

$$St = 0.029(Pr)^{-.4}(Re_x)^{-.2} \qquad (2\text{-}8)$$

where Re_x is the Reynolds number based on distance along the plate.

Also, for a flat plate,

$$\Delta_2 = \frac{.029}{.8Pr^{.4}}\left[\frac{\nu}{u_\infty}\right]^{.2}(x)^{.8} \qquad (2\text{-}9)$$

Using Equation 2-9, Equation 2-8 may be transformed to an expression with the Reynolds number based on enthalpy thickness so that the Stanton number may be expressed in terms of a local thickness parameter instead of plate length. This makes the Stanton number much less dependent on a pressure gradient which exists in the nozzle of this experiment. After this transformation, the expression becomes

$$St = 0.0125(Pr)^{-.5}(Re_{\Delta_2})^{-.25} \qquad (2\text{-}10)$$

In order to accommodate variations in free-stream velocity, density, and surface transverse radius for the nozzle, the integral energy equation for the local surface heat flux is introduced:

$$\frac{q}{c_p} = \frac{1}{r}\frac{d}{dx}\left[\Delta_2 r \rho_e u_e (T_s - T_{AW})\right] \qquad (2\text{-}11)$$

where r is nozzle radius and terms with subscript e are values at the edge of the boundary layer.

Combining Equations 2-10 and 2-11 and simplifying, using the fact that

$$h = (St)(c_p)(\rho_e u_e) = q/(T_s - T_{AW}) \qquad (2\text{-}12)$$

the following expression for heat transfer coefficient is obtained as in Reference 4, page 219:

$$h = \frac{.029(Pr^*)^{-.4}(r)^{.25}(g_c\mu^*)^{.2}\rho_e u_e c_p^*}{\left[\int_0^x (r)^{1.25}(\rho_e u_e)dx\right]^{.2}} \qquad (2\text{-}13)$$

The above quantities with asterisks are based on Eckert's reference temperature:

$$T^* = \frac{T_s + T_e}{2} + .22(T_{AW} - T_e) \qquad (2\text{-}14)$$

Having derived the theoretical method of heat transfer coefficient calculation, the details of the experimental arrangement and procedure will be discussed.

III. The Experiment

Mechanical Set-up

Nozzle. The nozzle (Figure 3) chosen for this experiment is a modified ASME nozzle [Reference 5]. The design was chosen such that the value of the flow coefficient would be as near to 1.0 as possible. The actual flow coefficient is 1.023. The throat diameter of the nozzle, 1.5 inches, was chosen such that choked flow could be achieved for at least 10 seconds, given the capacity of the laboratory's vacuum tanks. The throat is slightly elongated to allow the installation of gages. Once the throat diameter was calculated, the remainder of the nozzle dimensions were determined according to the specifications of Reference 5.

The nozzle was machined from aluminum stock. Aluminum was chosen due to its excellent conduction properties and low relative cost. After the nozzle is heated, temperature gradients quickly dissipate and uniform temperature is achieved. Four holes were bored and tapped at the nozzle throat in symmetric locations for instrumented bolts. Two holes, directly opposite each other, contained the heat gages, while the other two contained pressure taps. A tripwire was installed around the perimeter of the nozzle 0.8 inches upstream of the gages to insure turbulent flow.

Originally, the purpose of the rim on the nozzle was to facilitate cooling with liquid nitrogen to establish a

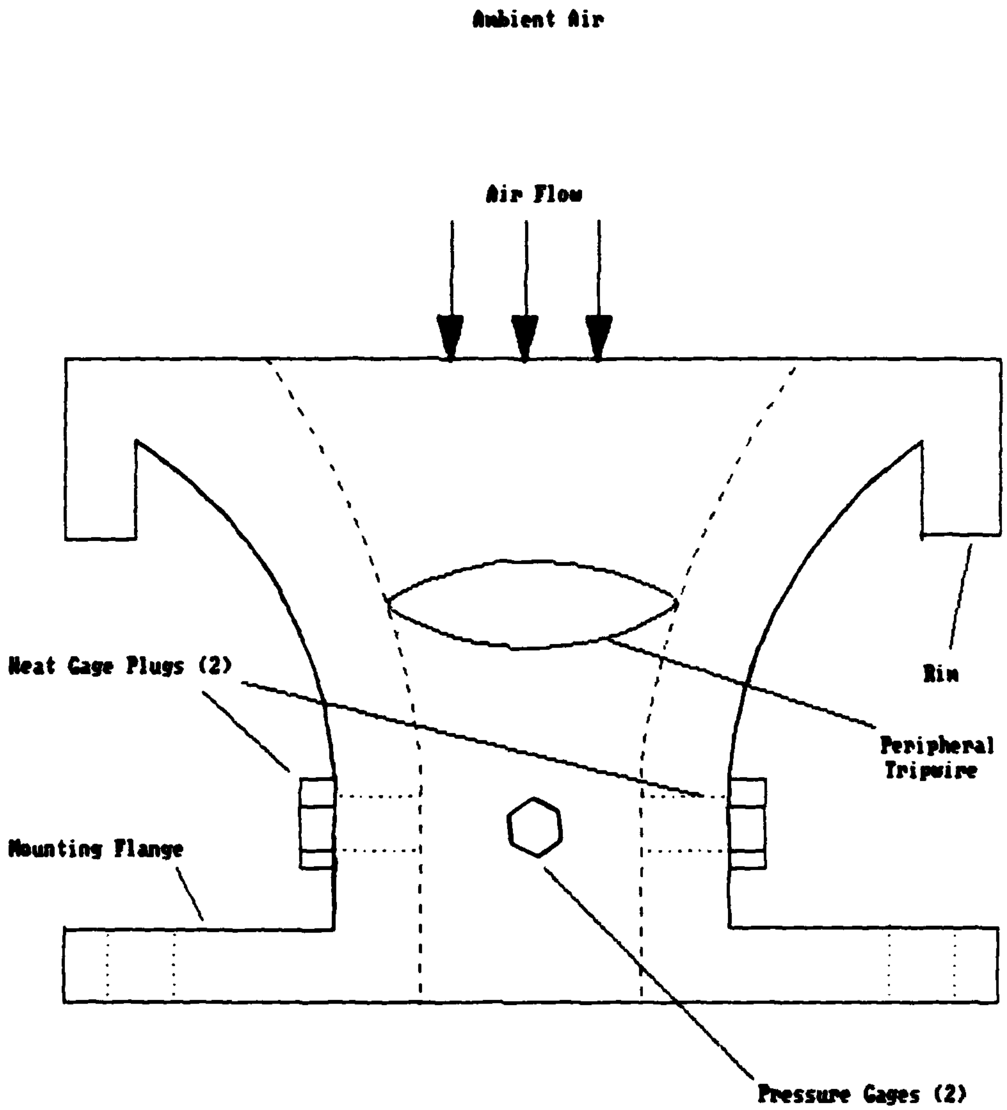

Figure 3: Nozzle and Gage Configuration

temperature difference between the nozzle and ambient air. The nozzle was mounted upside-down with respect to Figure 3 in order to prevent free convection inside the nozzle and associated piping. Liquid nitrogen was then poured directly onto the part of the nozzle circumscribed by the rim, enabling the nozzle to be cooled until the captured liquid was completely evaporated.

However, after several experimental runs, it became evident that mass transfer (condensation/evaporative cooling) was occuring. The liquid nitrogen reduced the nozzle temperature below the freezing point, allowing frost to form on the nozzle surface and increasing heat transfer to the surface. Then, later in the run, the frost melted and evaporated into the air stream, reducing heat transfer to the surface. Consequently, to avoid the mass transfer problem, heating was chosen as the method of establishing the temperature difference between the nozzle and ambient air. In order to prevent free convection with a heated nozzle, it was remounted as seen in Figure 3.

During evacuation, the nozzle was plugged by a standard number 12 rubber stopper with a bolt mounted in it. To initiate an experimental run, the plug was suddenly pulled from the nozzle with the aid of vise-grips.

Thin-film gages. Two thin-film gages (Figure 4) were used to gather data on heat transfer rates between the nozzle and the free stream. The gages, model number 1268, with serial numbers 2 and 2A, were made by TSI and consist

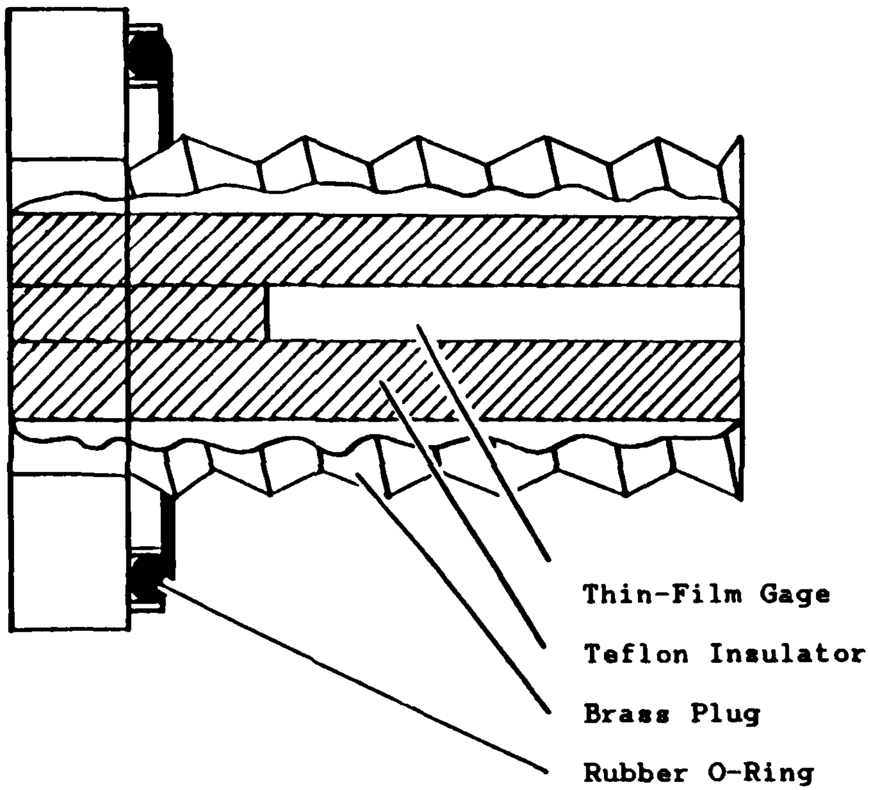

Figure 4: Thin-Film Gage in Brass Mounting Plug

of a thin strip of platinum which was vacuum deposited on a cylindrical quartz substrate. The substrate is 0.25 inch long and has a diameter of 0.125 inch. Each gage is mounted in a hollow Teflon tube of diameter 0.3875 inch, which, in turn, is mounted in a threaded brass plug. The nozzle is machined such that the center of the surface of each gage is flush with the inner wall of the nozzle when the brass plug is snugly inserted [Reference 2].

The gage resistance calibration from TSI was assumed to be correct. However, in order to verify that the gages were not damaged, a rough calibration was performed by heating the gages and comparing their bridge output with a digital thermometer as they cooled to room temperature. No anomalies were noted. Each gage has a unique resistance and rate of resistance change with temperature change. These were noted and accounted for in data reduction.

Pressure gages. Two Endevco 0-15 psia pressure gages were mounted opposite each other at the nozzle throat for the purpose of triggering the two oscilloscopes to initiate an experimental run. Given the purpose of the gages, and because the pressure change at the throat is nearly instantaneous when the rubber stopper is pulled, no precise calibration of the gages was performed.

However, it was still necessary to determine the pressure at the gage location in order to calculate the other flow properties. Since the gages were located in the elongated portion of the nozzle, upstream of the actual location of sonic flow, the flow at the gages was affected by friction. Therefore, knowing that the coefficient of friction of the flow is equal to $.029/(Re)^{.2}$ for turbulent flow [Reference 4, page 204-213], the Mach number at the gage location was found from the Fanno flow (flow with friction) portion of a standard set of gas tables [Reference 8]. This Mach number was then used to calculate the actual pressure at the location of the gages.

Vacuum system. The vacuum system consists of three vacuum pumps connected in parallel, sixteen interconnected vacuum tanks of appoximately 33.4 cubic feet each, and associated piping, giving a total capacity of over 535 cubic feet. Fifteen minutes are required to attain a pressure of approximately 0.1 psia in the system. This low pressure provides 35 seconds of sonic flow at the nozzle exit.

Instrumentation

Bridge/Amplifiers. The gages were connected to the bridge/amplifier circuits as illustrated in Figure 1. Four Hewlett-Packard PSC 8015-1 amplifiers were used to amplify the signals from the four gages (two for temperature, and two for pressure).

Digital Oscilloscopes. Two Tektronix 2430A digital oscilloscopes were used during each experimental run to record the thin-film heat transfer gage data from the amplifiers. Bridge output voltage was recorded as a function of time. Approximately 500 data plots were recorded during each run.

As testing progressed, it was noted that the oscilloscope readings were consistently varying from the readings of a digital voltmeter. Since the voltmeter was used to zero the gages and monitor drift during the equipment warm-up period, it was decided to use the voltmeter readings as the baseline. One oscilloscope was consistently adding 5.7% to the voltmeter reading and the other was subtracting 2.3%. Therefore, after each test, the data was adjusted accordingly.

After the correction was made, the data still contained a good deal of noise which gave erratic results. Therefore the data was used to obtain a best curve fit to a fifth degree polynomial. For any set of data, the correlation between the data and the curve was 98% or better. Data was then taken from these curves as input to the computer program QOE.FOR (see Appendix B), which solves Equations 2-3 and 2-4, to determine the heat transfer coefficient as a function of time.

Computer. Raw data recorded on the oscilloscopes was transferred to storage disks by a laboratory Zenith Z-248 computer. Discrete readings of the data (bridge output voltage versus time) were obtained using the computer and Tektronix software. Between 25 and 50 data points were used from each run to determine the experimental heat transfer coefficient, using the Fortran program QOE.FOR.

Experimental Procedure

Prior to an experimental run, the electrical components used for data recording were turned on and given 90 minutes to warm up. During warm-up time several things were accomplished. The bridges were balanced immediately after turning them on and were periodically checked for drift and rebalanced. The oscilloscope settings were initialized and ambient temperature and pressure were checked.

After warm-up and final bridge balancing were complete, the rubber plug was inserted in the nozzle and the system was evacuated to the capacity of the vacuum tank. During evacuation, a standard butane torch was used to heat the nozzle to approximately 150 degrees F. A digital voltmeter was used to indicate when the nozzle was hot enough. After heating, the voltmeter was used to read output from the two heat transfer gages to check for temperature gradients.

When the evidence of gradients was gone, the cables from the heat transfer gages were reconnected to the oscilloscopes and the oscilloscopes were set to trigger with a change of pressure gage output. The rubber stopper was then quickly pulled from the nozzle, withheld for the duration of run desired, and replaced in the nozzle. Data recorded by the oscilloscopes was immediately transferred to a computer disk, the oscilloscopes were reset and subsequent experimental runs were conducted. Because of the capacity of the vacuum system and short run times (10 seconds or less), it was possible to complete several runs without additional heating or pumping.

IV. Experimental Results

Data Reduction Anomalies

Two observations should be made concerning the method of data reduction. First, a few iterations of Equation 2-3 are required before sufficient numerical accuracy exists to get good results. Therefore, approximately the first four data points must not be accepted as the correct value. Second, the best curve fit method tended to skew the last two or three data points upward in most cases (see Figures 10 through 15 in Appendix A). However, these anomalies do not adversely affect the accomplishment of the objectives of this thesis, since the major area of interest is the point of divergence between theoretical and experimental values of the heat transfer coefficient, which in general does not occur at either endpoint of a data plot.

Specific Experimental Runs

The following five experimental runs were chosen to illustrate the accuracy of the gages. Runs #1, #2, and #5 were compiled from data taken from gage #2A, while Run #3 includes data from gage #2A and gage #2. Run #4 data was taken from gage #2.

Run #1. Figure 5 illustrates the characteristics of the gage output for a run of 0.1 second. The experimental range of the heat transfer coefficient h is shown to be consistently higher than the theoretically computed value of

0.0267 due to the effect of preheating. The maximum error caused by preheating (7.1%) occurs at t = 0.02 second. The relatively constant value of h indicates that the two-dimensional effect has not yet had a significant impact. Also evident are the data points at the beginning and end of the run which must be disregarded due to the data reduction anomalies (the first four points as well as the last three).

Run #2. Represented by Figure 6, this run is used to determine the point at which the experimental value of h diverges from the theoretical value. The two-dimensional effect is shown to be effective at t = 0.32 second, after which time the experimental and theoretical values of h diverge. However, if the 7.1% error due to preheating (seen above for Run #1) can be considered acceptable, run times can be extended to 0.60 second, where the error due to two-dimensional effects is approximately 7.2%. Allowable test time, then, is a function of accuracy desired. For example, if the error due to the two-dimensional effect is to be kept below 5%, gage readings should not be used past 0.5 second, where the error is 4.87%.

Run #3. This run points out the magnitude of error associated with relatively long experimental runs. Ignoring the last three data points of Figure 7, the data point at 8.8 seconds shows that the experimental result differs from the theoretical result by 52%. Lileikis' results at the end of a 4 second run [Reference 2, page 31] were erroneous by 42%, according to Figure 7. Run #3 also illustrates the

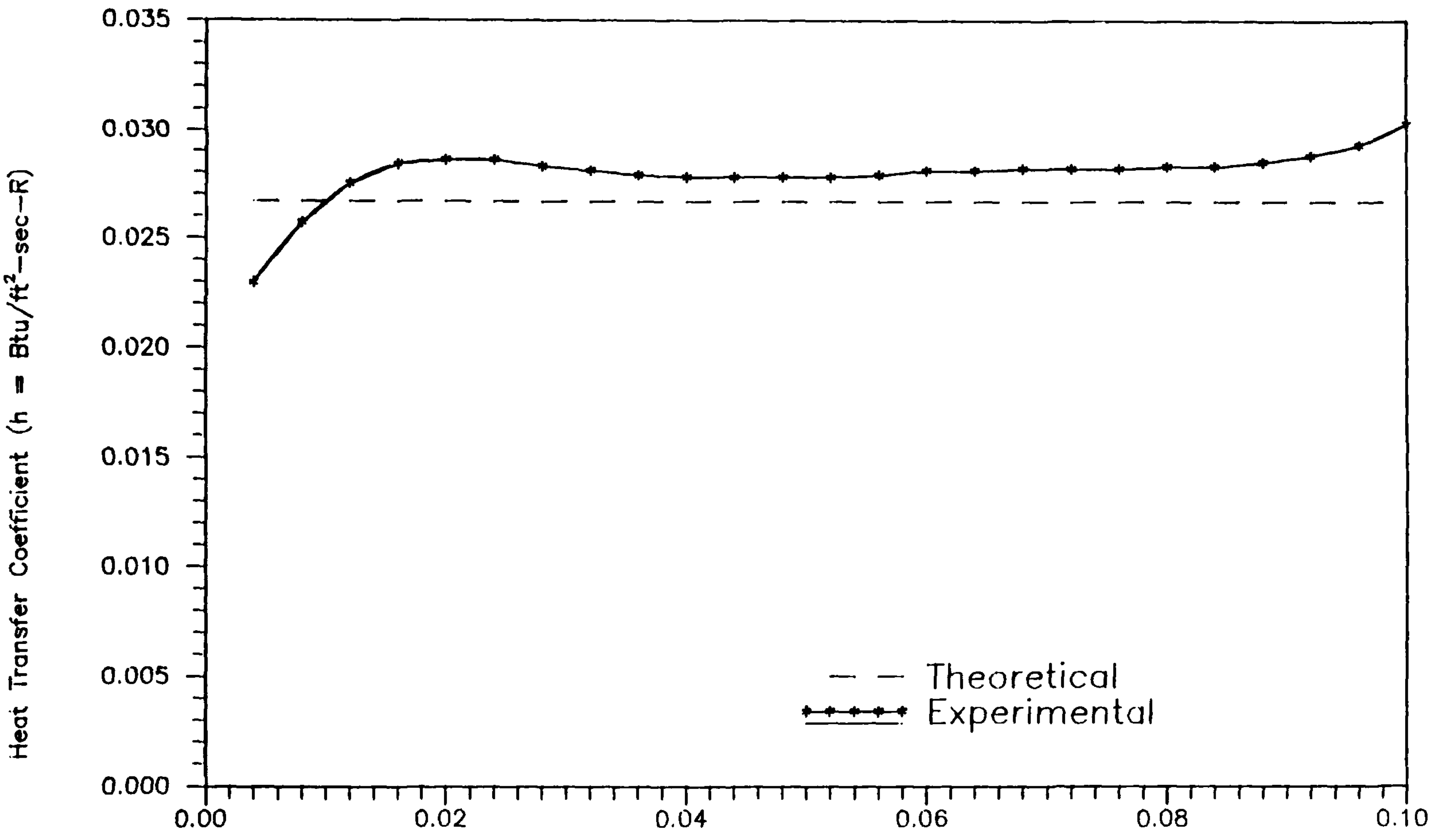

Figure 5: Run #1 – Heat Transfer Coefficient Versus Time

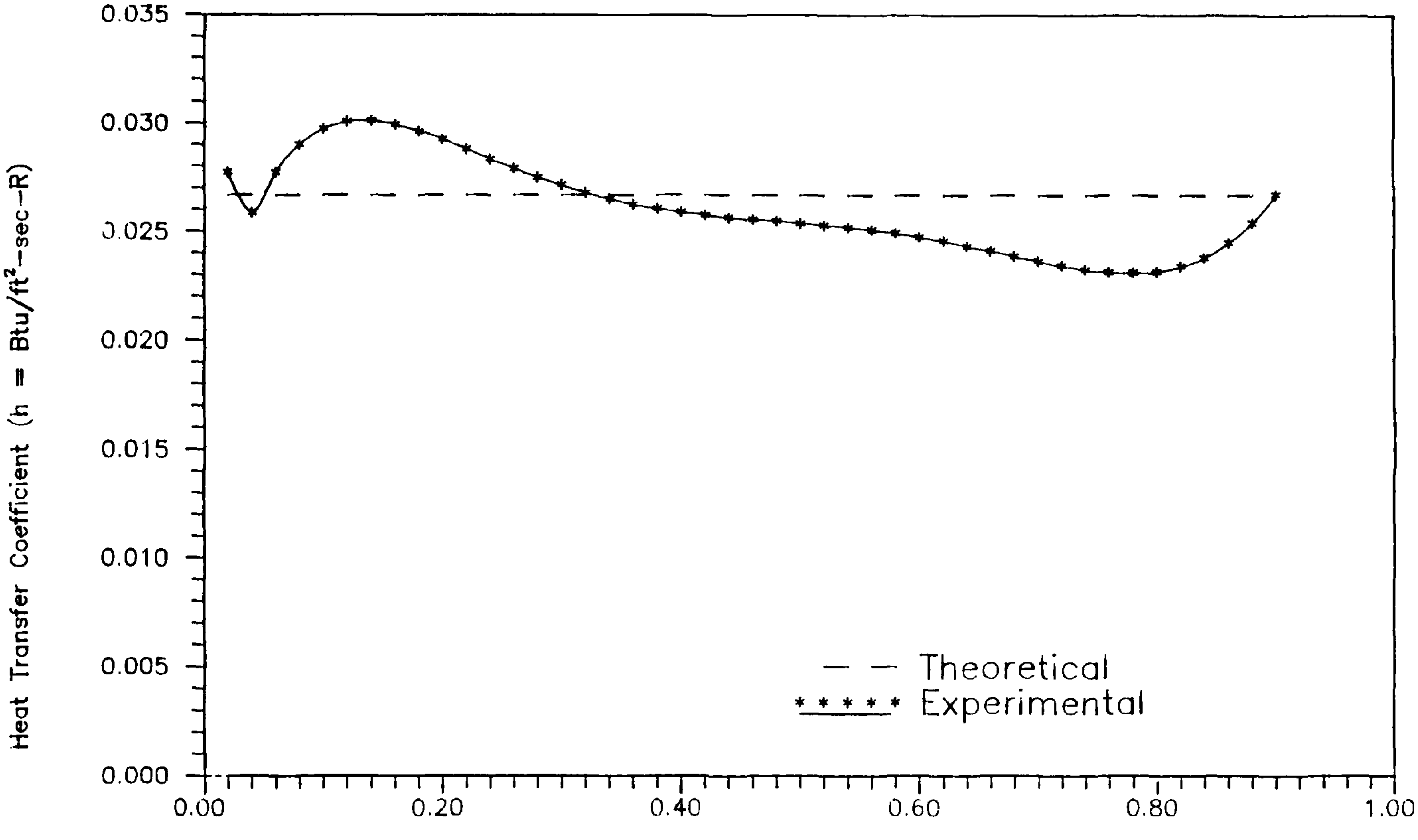

Figure 6: Run #2 – Heat Transfer Coefficient Versus Time

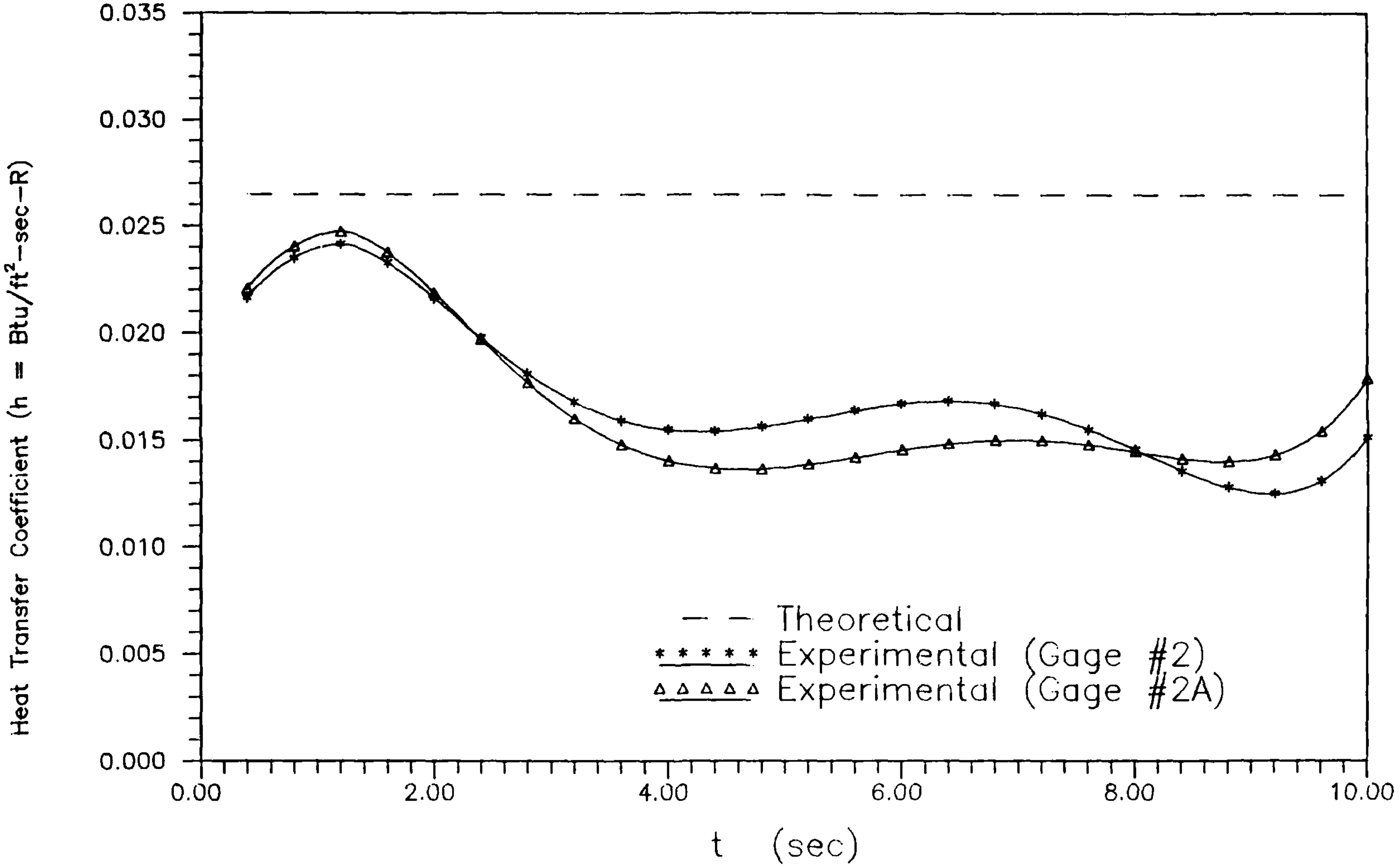

Figure 7: Run #3 – Heat Transfer Coefficient Versus Time

close resemblance between results from gage #2 and gage #2A, reinforcing confidence in the data taken.

Run #4. The effect of preheating due to an increased bridge voltage to 7.5 V is illustrated in Figure 8. The only difference between Run #4 and Run #3 is the different bridge voltages (5 V for Run #3). Therefore gage #2 data from Run #3 is included in Figure 8 for comparison. The voltage increase caused an increase of up to 27% in the value of h. Over the first 4 seconds of the run, it caused an average increase of 25% in the value of h. It can be seen that the increased effect of preheating persists throughout the run, as expected from the discussion of Section II. The preheating effect is diminished significantly in the latter stages of the run, where it is masked by the two-dimensional effect. However, by this time (5+ seconds into the run) gage readings are no longer accurate.

Run #5. This run was completed before any of the runs discussed previously. In the early stages of experimentation, the Reynolds number for an experimental run for average laboratory conditions was calculated to be 6.1×10^5, which indicated that the flow would be just into the turbulent range. However, after the initial runs were analyzed, the experimental value of h was only about 25% of the theoretical value based on turbulent flow. Therefore a theoretical analysis based on laminar flow was performed based on a derivation in Reference 4, page 155. This analysis is similar to that of Section II and the

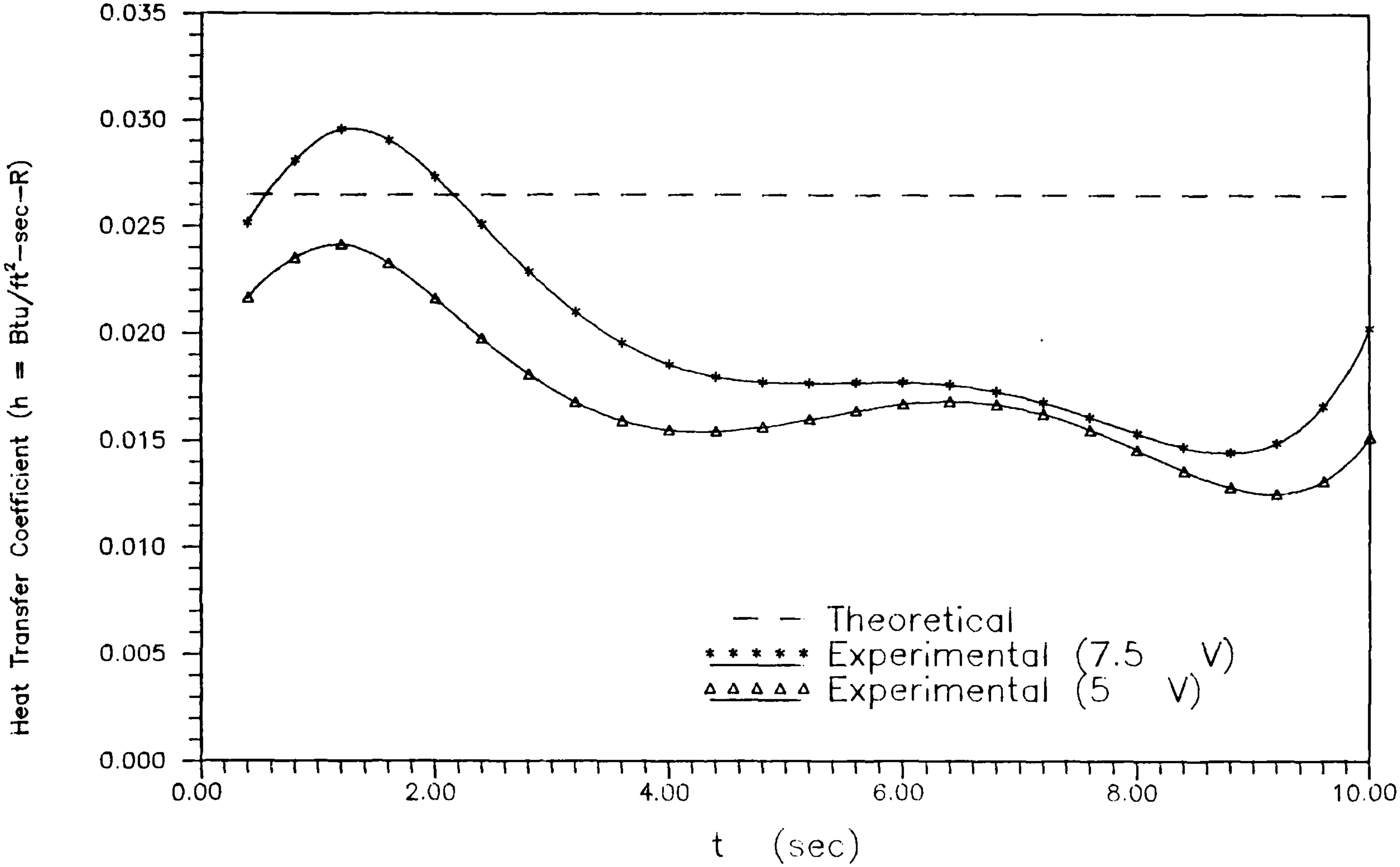

Figure 8: Run #4 — Effect of Increased Bridge Operating Voltage

computations were performed by QOL.FOR, a computer program included in the Appendix of this thesis. The laminar calculation agreed much more closely with the experimental results, indicating that the flow was laminar.

As seen in Figure 9, the experimental value of h exceeds the theoretical value until 0.52 second has elapsed. The most probable reason for the relatively long time for the two values to converge is that the increase in h due to an increase in surface temperature is more pronounced in laminar flow than turbulent flow [Reference 4]. Therefore, more time is required for the effects of preheating to be negated by two-dimensional effects in laminar flow. To ensure turbulence and to avoid the possibility of laminar or transition flow, the tripwire was installed in the nozzle for all subsequent runs.

Apart from two exceptions, two trends are evident in all the runs. First, the experimental value of the heat transfer coefficient h is greater than the theoretical value of h at the beginning of each run due to the preheating effect discussed in Section II. Second, the experimental h drops below the theoretical h at some point in each run and continues to decrease throughout the remainder of the run. This steady decrease is due to the two-dimensional effect of heat transfer also discussed in Section II. The exceptions occur in Runs #3 and #4, where the maximum axpected values of h would have occurred in the portion of the graph which

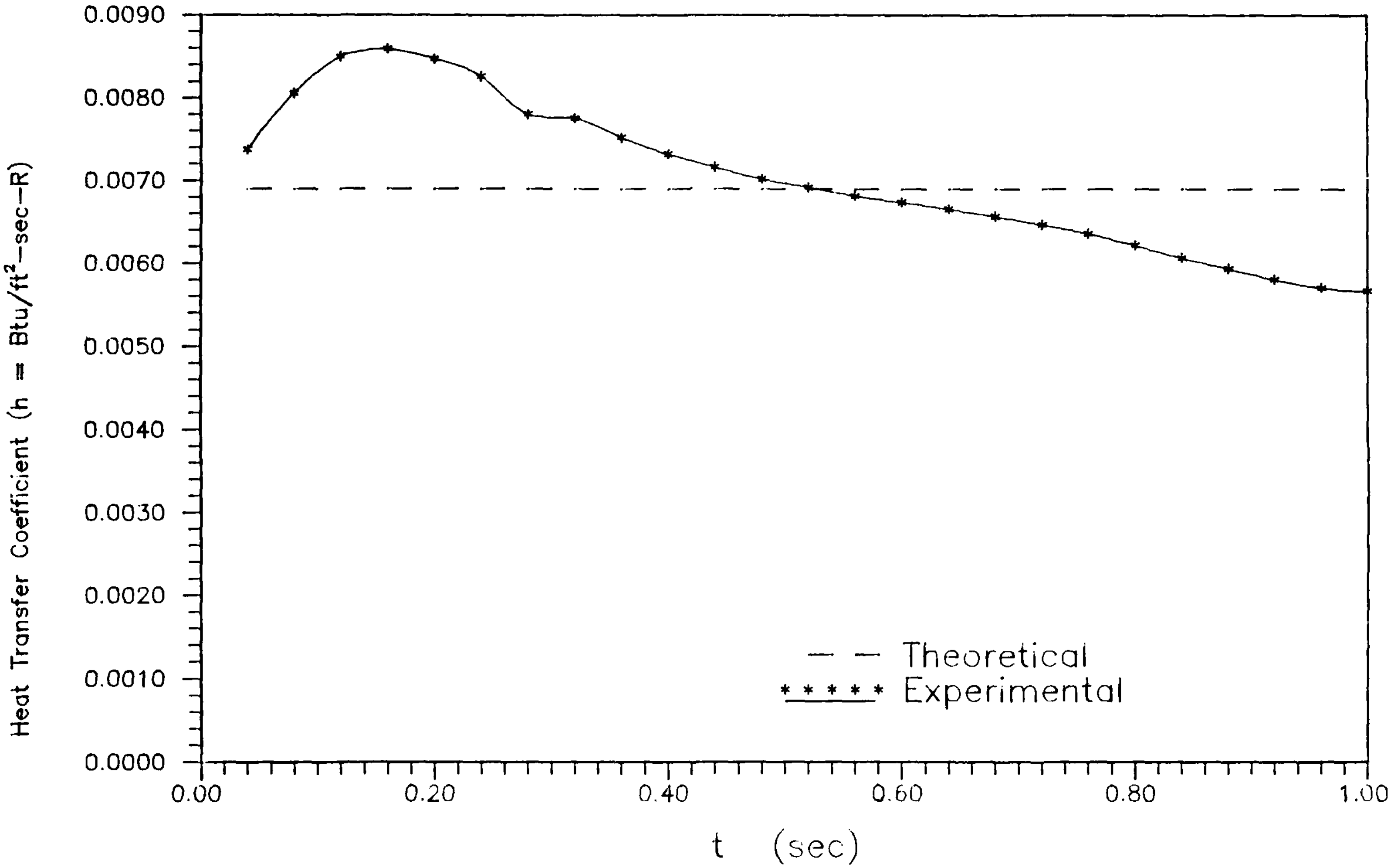

Figure 9: Run #5 - Laminar Flow Characteristics

is disregarded due to numerical inaccuracy, discussed previously.

V. Conclusions and Recommendations

Conclusions

Test time. Allowable test time is a function of accuracy desired. For example, if error in calculation of the heat transfer coefficient h due to two-dimensional effects is to be kept below 5%, experimental run time cannot exceed 0.5 second (see Run #2). On the other hand, if the error due to the two-dimensional effect is allowed to go as high as preheating error (7.1%), run time may be extended to 0.60 second.

Preheating effect. A 5 V bridge operating voltage caused a maximum error of 7.1% in the value of h for the flow conditions of this thesis. Although this appears significant, it may be acceptable for turbulent flow. The bridge voltage increase from 5 V to 7.5 V caused an average increase of 25% in the value of h. This points out the necessity of keeping bridge operating voltages as low as possible to avoid these effects.

Recommendations

As may be expected, the experiments represented by this thesis uncovered as many questions as answers. The following are recommendations for further study with thin-film gages.

Preheating effects. At least two things could be done to better understand this phenomenon. An experimental run should be accomplished with each gage operating at a different bridge input voltage. A single run provides identical conditions for the two gages, giving an improved basis for comparison. Also, instead of small voltage changes, experiments should be run with large changes in bridge input voltages, possibly allowing the heat generation flux due to the operating current to equal up to half the surface heat flux due to air flow in the nozzle.

Mass transfer. Given the conditions discussed in Section III, in which condensation and evaporative cooling resulted from the use of liquid nitrogen to cool the nozzle, a study of mass transfer could be accomplished using the knowledge about thin-film gages gleaned from this thesis.

Two-dimensional effect minimization. In order to minimize the two-dimensional effect, experimentation could be done with gage substrates with a thermal diffusivity similar to aluminum. Another way to minimize the two-dimensional effect is to use a better material than Teflon to insulate the gage from the nozzle.

Laminar flow. Further experimentation with laminar flow should be accomplished to better understand the difference between laminar and turbulent boundary layers' ability to adjust to nonisothermal surfaces.

Appendix A: Raw Data and Curve Fits

Run #1 (Gage #2A)

Stagnation temperature: 533.4 R
Stagnation pressure: 14.364 psia
Initial heated nozzle temperature (TONE): 579 R

Elapsed experimental time Δt in seconds versus change of bridge output voltage ΔV in millivolts.

Δt	ΔV
0.0040	0.0800
0.0080	0.0800
0.0120	0.1200
0.0160	0.1600
0.0200	0.1600
0.0240	0.2000
0.0280	0.2000
0.0320	0.2400
0.0360	0.2400
0.0400	0.2400
0.0440	0.2800
0.0480	0.2800
0.0520	0.2800
0.0560	0.2800
0.0600	0.3200
0.0640	0.2800
0.0680	0.3200
0.0720	0.3600
0.0760	0.3600
0.0800	0.3600
0.0840	0.3600
0.0880	0.3600
0.0920	0.3600
0.0960	0.4000
0.1000	0.4000

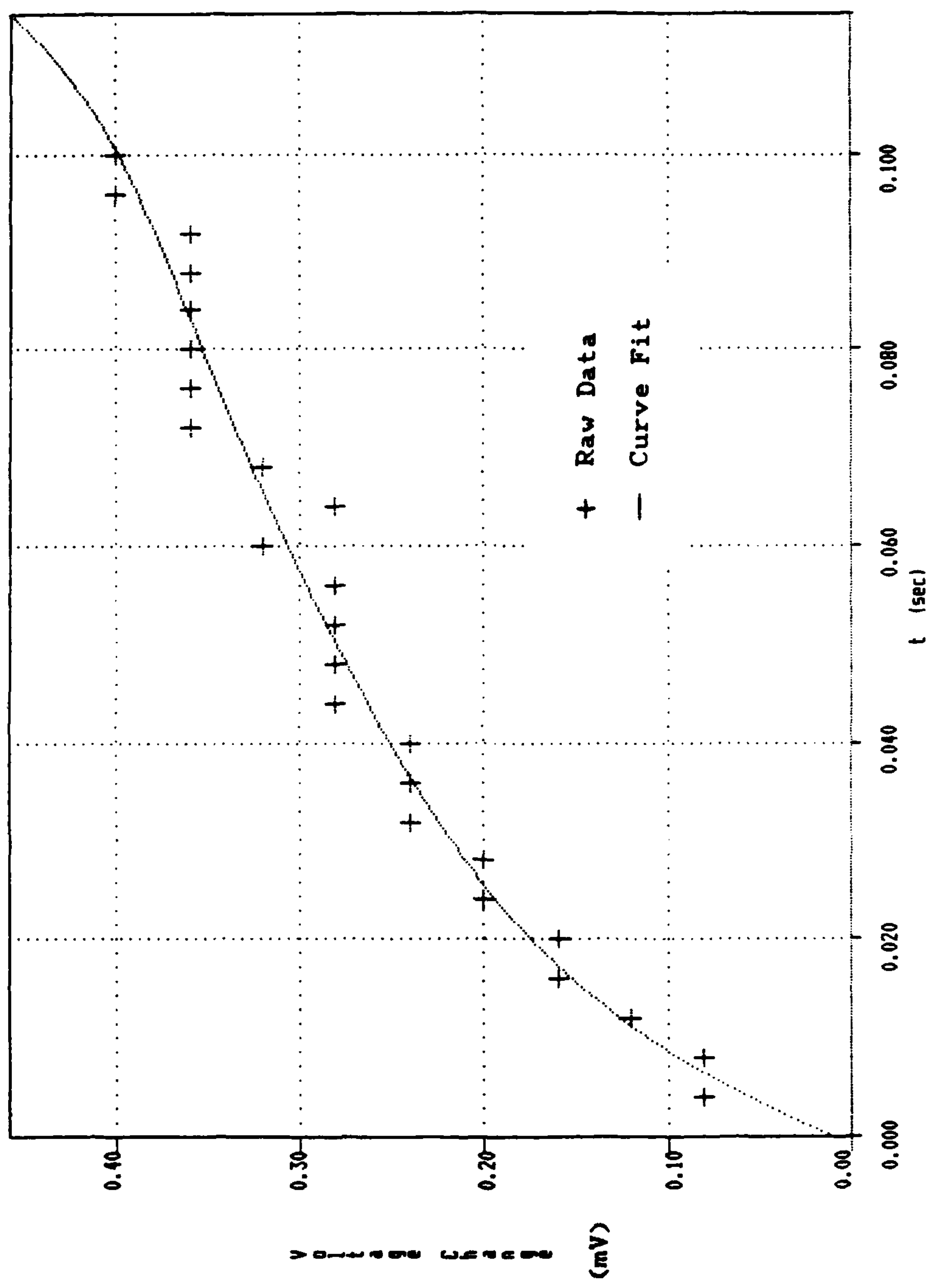

Figure 10: Curve Fit (Run #1, Gage #2A)

Run #2 (Gage #2A)

Stagnation temperature: 553.4 R
Stagnation pressure: 14.364 psia
Initial heated nozzle temperature: 579 R

Δt	ΔV
0.0200	0.1600
0.0400	0.2800
0.0600	0.3200
0.0800	0.3600
0.1000	0.3600
0.1200	0.4000
0.1400	0.4400
0.1600	0.4800
0.1800	0.5200
0.2000	0.5200
0.2200	0.5600
0.2400	0.6000
0.2600	0.6000
0.2800	0.6400
0.3000	0.6400
0.3200	0.6400
0.3400	0.6400
0.3600	0.6800
0.3800	0.6800
0.4000	0.7200
0.4200	0.7200
0.4400	0.7200
0.4600	0.7200
0.4800	0.7600
0.5000	0.7600
0.5200	0.7600
0.5400	0.7600
0.5600	0.8000
0.5800	0.8000
0.6000	0.8000
0.6200	0.8000
0.6400	0.8000
0.6600	0.8000
0.6800	0.8000
0.7000	0.8000

0.7200	0.8400
0.7400	0.8400
0.7600	0.8800
0.7800	0.8800
0.8000	0.8800
0.8200	0.8800
0.8400	0.8800
0.8600	0.8800
0.8800	0.8800
0.9000	0.9200

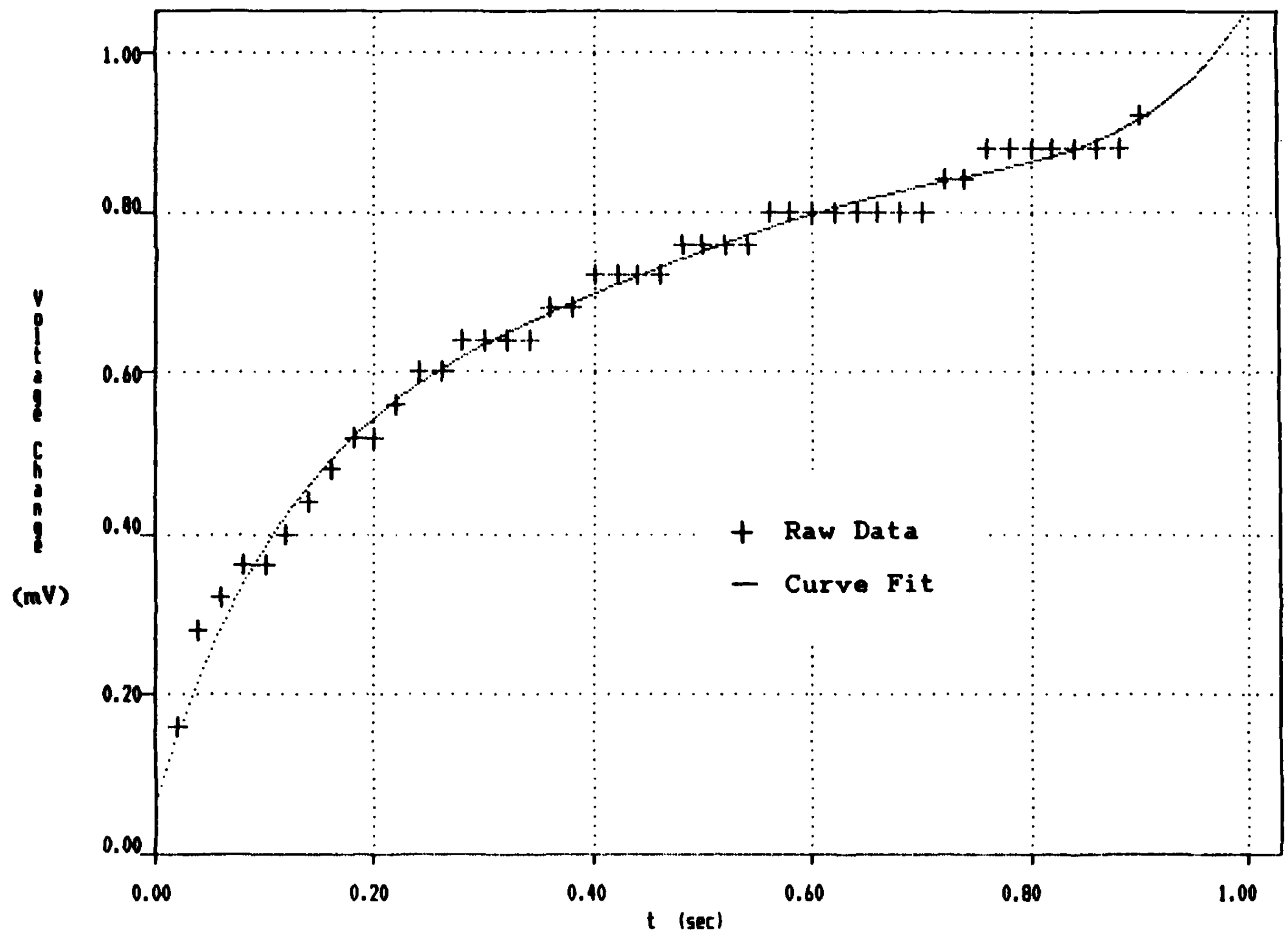

Figure 11: Curve Fit (Run #2, Gage #2A)

Run #3 (Gage #2)

Stagnation temperature: 528.9 R
Stagnation pressure: 14.229 psia
Initial heated nozzle temperature: 571.8 R

Δt	ΔV
0.4000	0.5000
0.8000	0.8000
1.2000	0.9000
1.6000	0.9000
2.0000	1.0000
2.4000	1.0000
2.8000	1.1000
3.2000	1.1000
3.6000	1.2000
4.0000	1.2000
4.4000	1.2000
4.8000	1.2000
5.2000	1.3000
5.6000	1.3000
6.0000	1.3000
6.4000	1.3000
6.8000	1.3000
7.2000	1.3000
7.6000	1.4000
8.0000	1.4000
8.4000	1.4000
8.8000	1.4000
9.2000	1.4000
9.6000	1.4000
10.0000	1.4000

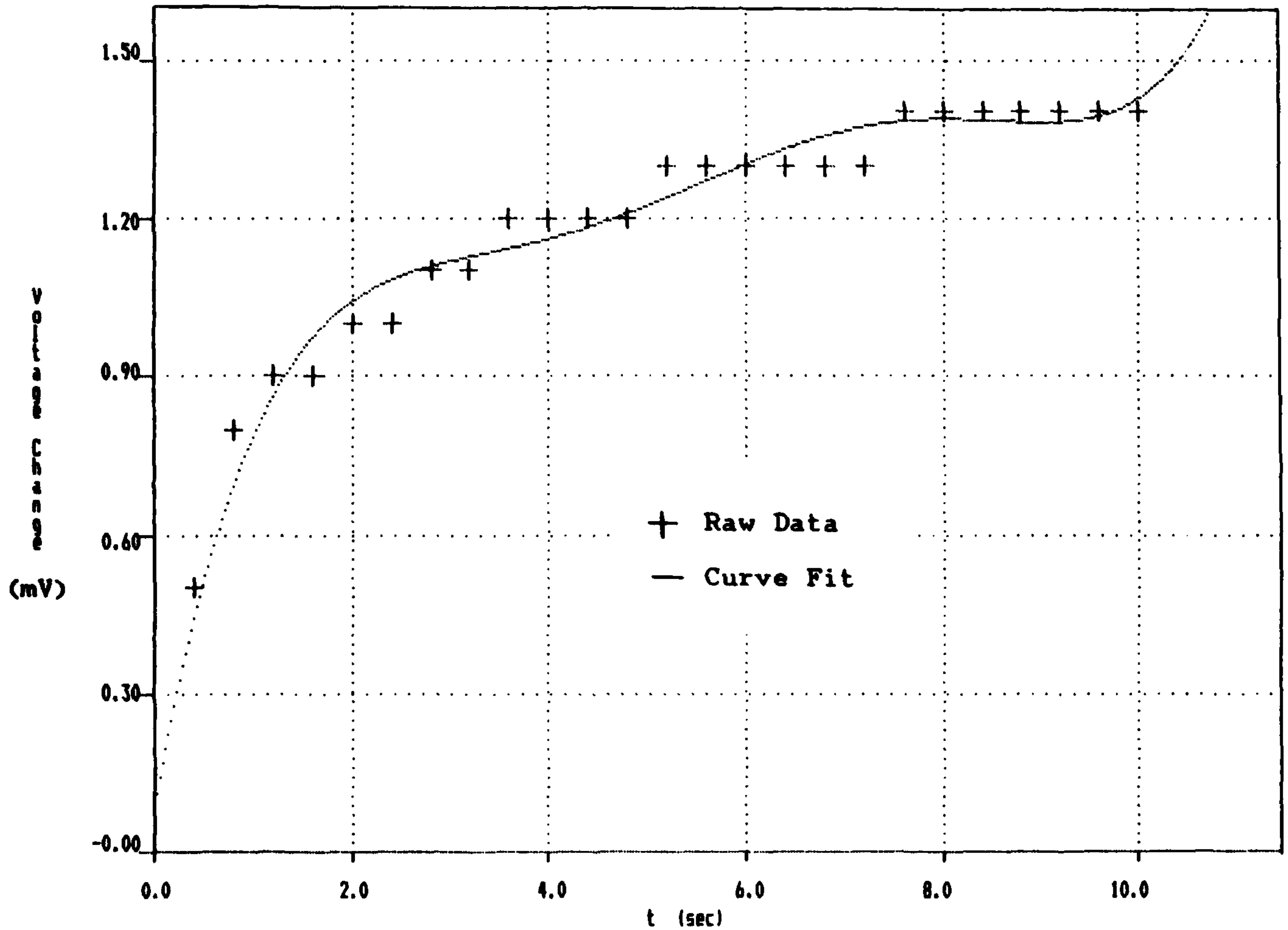

Figure 12: Curve Fit (Run #3, Gage #2)

<u>Run #3</u> (Gage #2A)

Stagnation temperature: 528.9 R
Stagnation pressure: 14.229 psia
Initial heat nozzle temperature: 573.2 R

Δt	ΔV
0.4000	0.5000
0.8000	0.8000
1.2000	0.9000
1.6000	0.9000
2.0000	1.0000
2.4000	1.0000
2.8000	1.1000
3.2000	1.1000
3.6000	1.1000
4.0000	1.1000
4.4000	1.2000
4.8000	1.2000
5.2000	1.2000
5.6000	1.2000
6.0000	1.2000
6.4000	1.2000
6.8000	1.2000
7.2000	1.3000
7.6000	1.3000
8.0000	1.3000
8.4000	1.3000
8.8000	1.4000
9.2000	1.4000
9.6000	1.4000
10.0000	1.4000

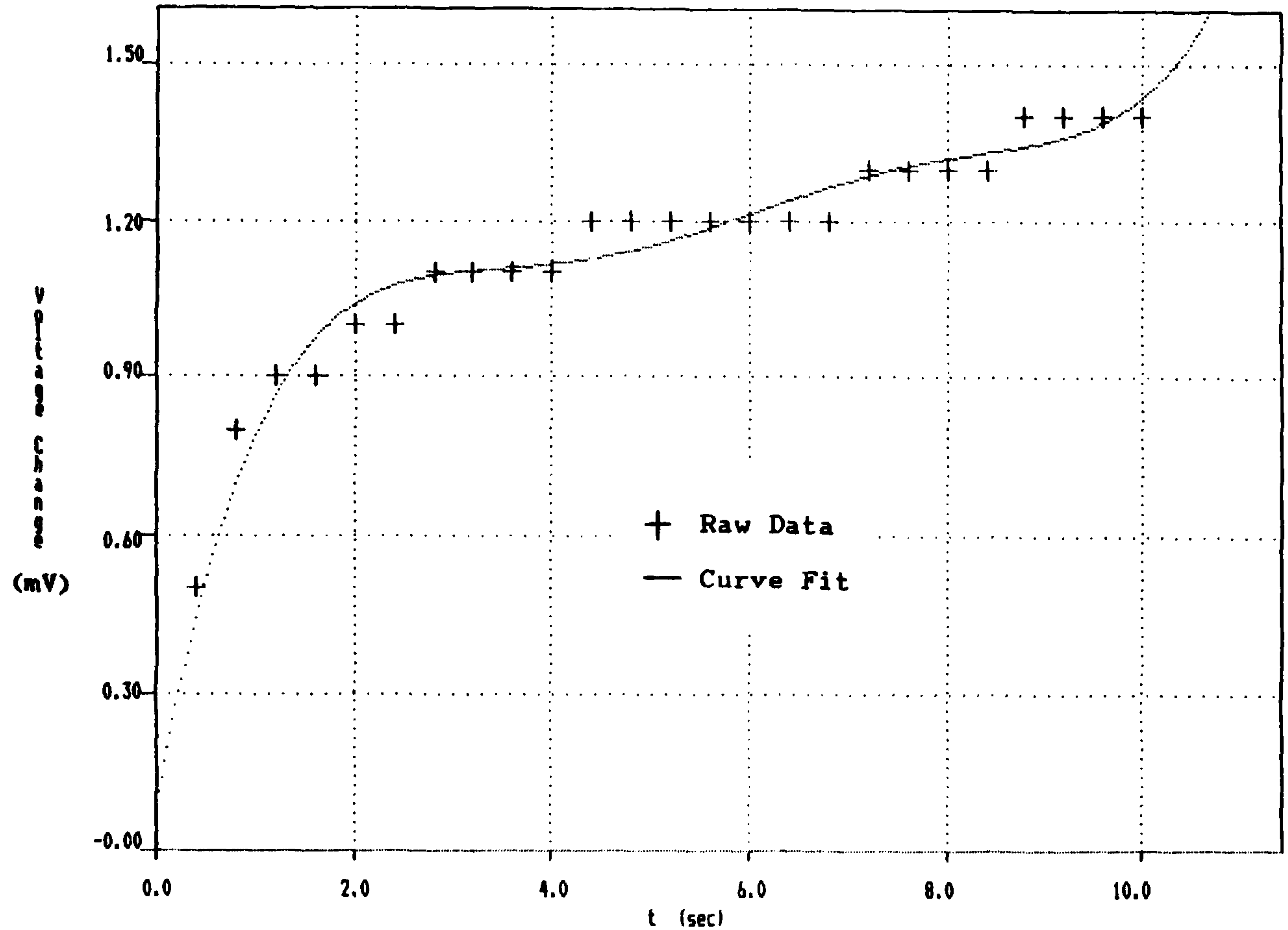

Figure 13: Curve Fit (Run #3, Gage #2A)

Run #4 (Gage #2)

Stagnation temperature: 530.7 R
Stagnation pressure: 14.393 psia
Initial heated nozzle temperature: 566.7 R

Δt	ΔV
0.4000	0.8000
0.8000	1.1000
1.2000	1.3000
1.6000	1.4000
2.0000	1.5000
2.4000	1.6000
2.8000	1.6000
3.2000	1.7000
3.6000	1.7000
4.0000	1.8000
4.4000	1.8000
4.8000	1.8000
5.2000	1.8000
5.6000	1.8000
6.0000	1.8000
6.4000	1.8000
6.8000	1.9000
7.2000	1.9000
7.6000	1.9000
8.0000	1.9000
8.4000	1.9000
8.8000	1.9000
9.2000	2.0000
9.6000	2.0000
10.0000	2.0000

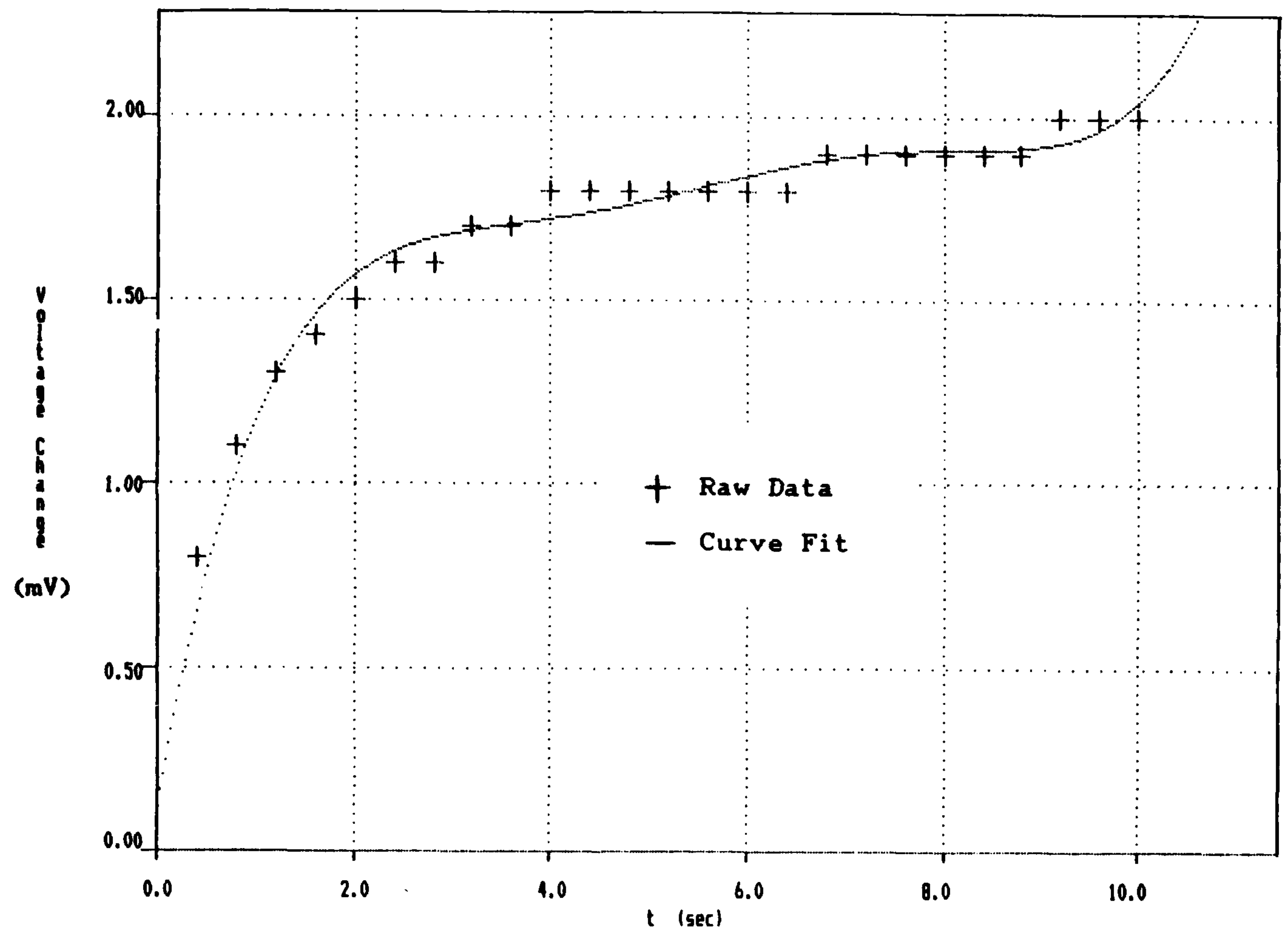

Figure 14: Curve Fit (Run #4, Gage #2)

Run #5 (Gage #2A)

Stagnation temperature: 533.4 R
Stagnation pressure: 14.325 psia
Initial heated nozzle temperature: 592.2 R

Δt	ΔV
0.0400	0.0800
0.0800	0.1200
0.1200	0.1600
0.1600	0.1600
0.2000	0.2000
0.2400	0.2400
0.2800	0.2400
0.3200	0.2400
0.3600	0.2400
0.4000	0.2800
0.4400	0.2800
0.4800	0.2800
0.5200	0.3200
0.5600	0.3200
0.6000	0.3200
0.6400	0.3200
0.6800	0.3200
0.7200	0.3200
0.7600	0.3200
0.8000	0.3600
0.8400	0.3600
0.8800	0.3600
0.9200	0.3200
0.9600	0.3600
1.0000	0.3600

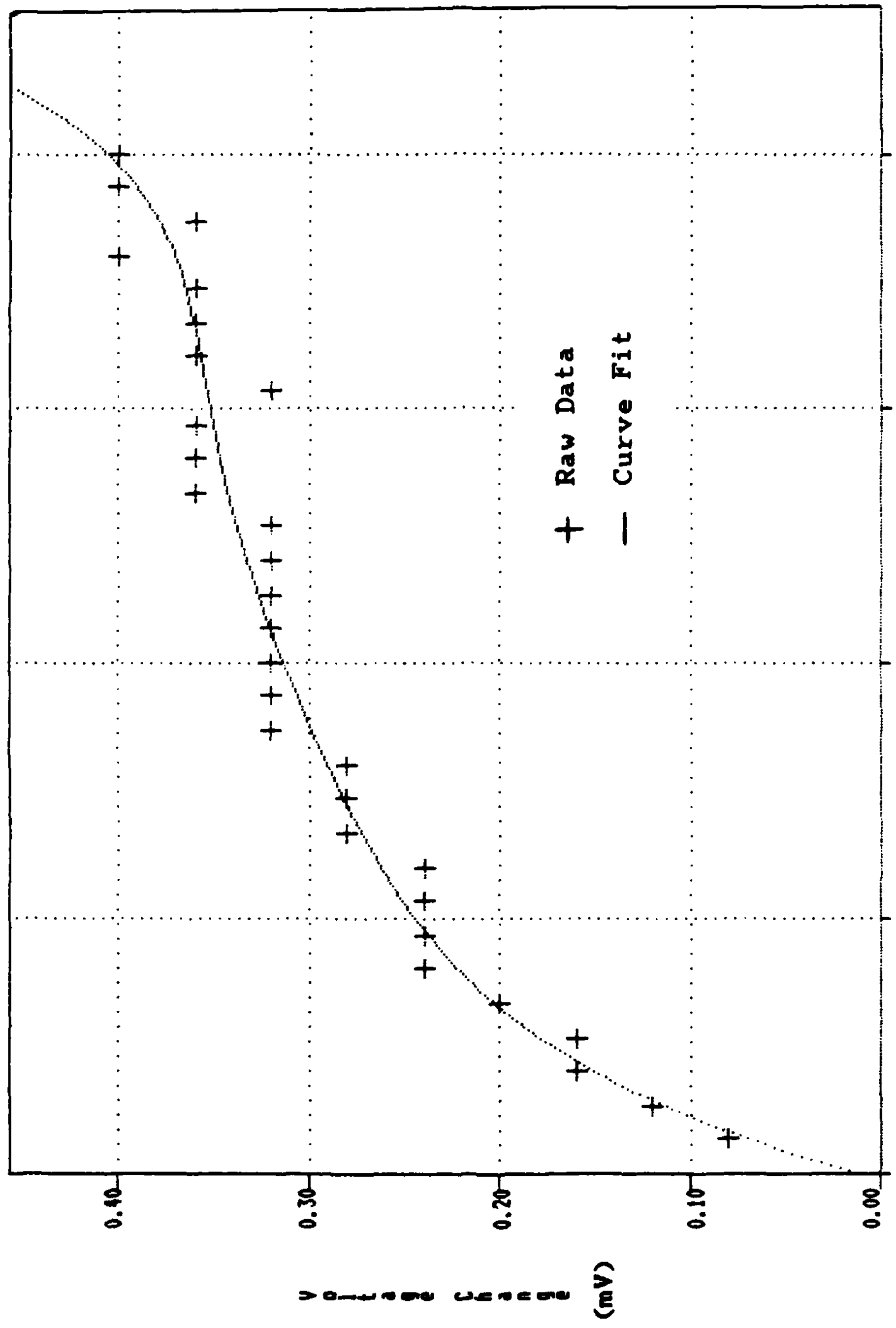

Figure 15: Curve Fit (Run #5, Gage #2A)

Appendix B: Computer Programs

```
************************************************************
*                                                          *
*                        QOT.FOR                           *
*                                                          *
*   Solving for the turbulent theoretical value of the     *
*  heat transfer coefficient between the free stream and   *
*   the nozzle, based on Equation 12-32 in Reference 4.    *
*                                                          *
************************************************************
C
C   First, input the initial wall temperature, TONE.
      WRITE(*,9)'Please input initial wall temperature,TONE'
      WRITE(*,9)'Note: use exponential notation'
9     FORMAT(1X,A)
      READ(*,1) TONE
1     FORMAT(1E14.7)
C
C   Now prompt for room temperature, TO
C
      WRITE(*,9)'Please input room temperature,TO'
      READ(*,1) TO
C
C   Now prompt for the adiabatic wall temperature, TAW
C
      WRITE(*,9)'Please input the adiabatic wall temp, TAW'
      READ(*,1) TAW
C
C   Now prompt for the ambient pressure, PO
C
      WRITE(*,9)'Please input the ambient pressure, PO'
      READ(*,1) PO
C
C   Now solve for the edge temperature, TE
C
      TE=.83333*TO
C
C   Find the reference temperature, TEMSTR
C
      TEMSTR=((TONE+TE)/2)+(.22)*(TAW-TE)
C
C   Solve for the correct value of viscosity, GCMU
C
      DT=TEMSTR-450.0
      DMU=DT*34.0/200.0
      GCMU1=DMU+109.0
      GCMU=(GCMU1)*(1.0E-07)/12.0
C
C   List other values in numerator of eqn. 12-32
C
      PR=.71
      R=.75
```

```
      FPAREA=.5297*PO/SQRT(TO)
      CPE=.24
      DELTAT=TONE-TAW
C
C   Now solve for the numerator (TOP) of 12-32
C
      A1=(.029)*(PR**(-.4))*(R**(.25))*(GCMU**(.2))
      A2=(FPAREA)*(CPE)*(DELTAT)
      TOP=A1*A2
C
C   Now solve for the denominator, using Simpson's method
C
      I=1
      SUM=0.0
      FLOW=0.5999
      F1=((1.5)**1.25)*((FLOW)/((3.141593*(1.5)**2)))
      X=1.49
      STOT=0.0
      Y1=0.0
      DLTAX2=.0001
      WRITE(*,6)
6     FORMAT(5X,'S1',9X,'STOT',7X,'R',10X,'A',9X,'AREA',
     *7X,'SUM')
2     Y2=SQRT(1-(X/1.5)**2)*(.75)
      DELTAY=Y2-Y1
      DLTAY2=DELTAY**2
      S1=SQRT(DLTAY2+DLTAX2)
      STOT=STOT+S1
      R=1.5-Y2
      A=(3.141593)*(R**2)
      F2=(R**1.25)*(FLOW/A)
      AREA=(F1+F2)*(S1/2)
      SUM=SUM+AREA
      F1=F2
4     WRITE(*,5) S1,STOT,R,A,AREA,SUM
      IF (I.EQ.2) THEN
            GO TO 7
          ELSE
            GO TO 8
      END IF
8     Y1=Y2
      X=X-.01
      IF (X.LT.0.0) THEN
            GO TO 3
          ELSE
            GO TO 2
      END IF
3     R=0.75
      A=1.76715
      S1=0.185
      STOT=STOT+S1
      AREA=(R**1.25)*(FLOW/A)*S1
      SUM=SUM+AREA
      I=2
```

```
      GO TO 4
C
C   Now solve for total heat transfer per second-foot-squared
C   Multiply by 144 to convert from sq. in. to sq. ft.
C
7     QO=(TOP*144.0)/(SUM**(.2))
C
C   Now solve for the heat transfer coefficient
C
      AITCH=QO/(TONE-TAW)
C
      WRITE(*,10) QO
      WRITE (*,11) AITCH
10    FORMAT(//' QO = ',1E14.7,' Btu/ft-squared-second')
11    FORMAT(' h = ',1E14.7,'Btu/ft-squared-sec-degR')
      STOP
5     FORMAT(1X,6(F8.5,3X))
      END
```

```
*************************************************************
*                                                           *
*                        QOE.FOR                            *
*                                                           *
*        Solving for the heat transfer coefficient          *
*       between the free stream and the nozzle wall,        *
*               based on experimental data.                 *
*                                                           *
*************************************************************
C
C   Declare the time and voltage arrays, TIME and VOLT.
C
      REAL TIME(50), VOLT(50)
C
C   Prompt for the adiabatic wall temperature, TAW
C
      WRITE(*,1)'Please input adiabatic wall temp, TAW'
      READ(*,4) TAW
C
C   Prompt for the initial hot nozzle temp, TONE.
C
      WRITE(*,1)'Please input initial hot noz temp, TONE'
      READ(*,4) TONE
C
C   Prompt for the constant which converts voltages to
C   temperatures, CONST.
C
      WRITE(*,1)'Please input the constant multiplier, CONST'
      READ(*,4) CONST
C
C   Prompt for number of data points, J.
C
      WRITE(*,1)'Please input number of data points'
      READ(*,12) J
C
C   Prompt for time and voltage data inputs.
C
      WRITE(*,1)'Input time data exponentially'
      READ(*,2) (TIME(I),I=1,J)
1     FORMAT(1X,A)
2     FORMAT(5E14.7)
      I=1
15    WRITE(*,4) TIME(I)
      I=I+1
      IF(I.GT.J) THEN
          GO TO 13
        ELSE
          GO TO 15
      ENDIF
C
13    WRITE(*,1)'Input voltage data exponentially'
      READ(*,2) (VOLT(I),I=1,J)
C
C   Enter the value for A, composed of various property
```

```
C    values of the quartz substrate.
C
       A=.0832
C
C    Calculate heat transfer for N=1
C
       GSUM=0.0
       N=1
       I=1
       B=(TO-TO)/(2*((TIME(N))**(.5)))
       C=VOLT(I)/CONST
       D=0.0
       E=0.0
       F=(TIME(N))**(.5)
       G=(C-D)/(E+F)
       QO=A*(B+G)
       TW=TONE-C
       AITCH=QO/(TW-TAW)
       WRITE(*,5) TIME(N)
       WRITE(*,6) QO
       WRITE(*,3) AITCH
C
C    Now calculate the heat transfer for N=2,...,J
C
       N=2
7      B=(TO-TO)/(2*((TIME(N))**(.5)))
       I=1
       C=VOLT(I)/CONST
       D=0.0
       E=(TIME(N)-TIME(I))**(.5)
       F=(TIME(N))**(.5)
       G=(C-D)/(E+F)
       GSUM=GSUM+G
          I=2
9         C=VOLT(I)/CONST
          D=VOLT(I-1)/CONST
          E=(TIME(N)-TIME(I))**(.5)
          F=(TIME(N)-TIME(I-1))**(.5)
          G=(C-D)/(E+F)
          GSUM=GSUM+G
          I=I+1
          IF (I.GT.N) THEN
                GO TO 8
             ELSE
                GO TO 9
          ENDIF

8      QO=A*(B+GSUM)
       TW=TONE-C
       AITCH=QO/(TW-TAW)
       WRITE(*,5) TIME(N)
       WRITE(*,6) QO
       WRITE(*,3) AITCH
       N=N+1
```

```
      GSUM=0.0
      IF(N.GT.J) THEN
           GO TO 10
         ELSE
           GO TO 7
      ENDIF
10    STOP
4     FORMAT(1E14.7)
5     FORMAT(//' AT TIME = ',1E14.7,' SECONDS')
6     FORMAT(' HEAT TRANSFER = ',1E14.7,' BTU/FT-SQUARED-SEC')
3     FORMAT(' h = ',1E14.7,' BTU/FT-SQUARED-SEC-degR')
12    FORMAT(I2)
      END
```

```
**************************************************************
*                                                            *
*                          QOL.FOR                           *
*                                                            *
*     Solving for the theoretical laminar heat transfer      *
*     coefficient between the free stream and the nozzle     *
*        wall, based on Equation 9-50 in Reference 4.        *
*                                                            *
**************************************************************
C
C   First, input the initial wall temperature, TONE.
      WRITE(*,9)'Please input initial wall temperature,TONE'
      WRITE(*,9)'Note: use exponential notation'
9     FORMAT(1X,A)
      READ(*,1) TONE
1     FORMAT(1E14.7)
C
C   Now prompt for room temperature, TO
C
      WRITE(*,9)'Please input room temperature,TO'
      READ(*,1) TO
C
C   Now prompt for the adiabatic wall temperature, TAW
C
      WRITE(*,9)'Please input the adiabatic wall temp, TAW'
      READ(*,1) TAW
C
C   Now prompt for the ambient pressure, PO
C
      WRITE(*,9)'Please imput the ambient pressure, PO'
      READ(*,1) PO
C
C   Now solve for the edge temperature, TE
C
      TE=.83333*TO
C
C   Find the reference temperature, TEMSTR
C
      TEMSTR=((TONE+TE)/2)+(.22)*(TAW-TE)
C
C   Solve for the correct value of viscosity, GCMU
C
      DT=TEMSTR-450.0
      DMU=DT*34.0/200.0
      GCMU1=DMU+109.0
      GCMU=(GCMU1)*(1.0E-07)/12.0
C
C   List other values in numerator of eqn. 12-32
C
      PR=.71
      R=.75
      FPAREA=.5297*PO/SQRT(TO)
      CPE=.24
      DELTAT=TONE-TAW
```

```
C
C    Now solve for the numerator (TOP) of 12-32
C
       A1=(.418)*(R)*(GCMU**(.5))
       A2=(FPAREA**(1.435))*(CPE)*(DELTAT)
       TOP=A1*A2
C    Now solve for the denominator, using Simpson's method
C
       I=1
       SUM=0.0
       FLOW=.5802
       F1=((1.5)**2.)*(((FLOW)/(3.141593*(1.5**2.)))**1.87)
       X=1.49
       STOT=0.0
       Y1=0.0
       DLTAX2=.0001
       WRITE(*,6)
6      FORMAT(5X,'S1',9X,'STOT',7X,'R',10X,'A',9X,'AREA',
     *7X,'SUM')
2      Y2=SQRT(1-(X/1.5)**2)*(.75)
       DELTAY=Y2-Y1
       DLTAY2=DELTAY**2
       S1=SQRT(DLTAY2+DLTAX2)
       STOT=STOT+S1
       R=1.5-Y2
       A=(3.141593)*(R**2)
       F2=(R**2.)*((FLOW/A)**(1.87))
       AREA=(F1+F2)*(S1/2)
       SUM=SUM+AREA
       F1=F2
4      WRITE(*,5) S1,STOT,R,A,AREA,SUM
       IF (I.EQ.2) THEN
             GO TO 7
           ELSE
             GO TO 8
       END IF
8      Y1=Y2
       X=X-.01
       IF (X.LT.0.0) THEN
             GO TO 3
            ELSE
             GO TO 2
       END IF
3      R=.75
       A=1.76715
       S1=.185
       STOT=STOT+S1
       AREA=(R**2.)*((FLOW/A)**(1.87))*S1
       SUM=SUM+AREA
       I=2
       GO TO 4
C
C    Now solve for total heat transfer per second-foot-squared
C    Multiply by 144 to convert from sq. in. to sq. ft.
```

```
C
7       QO=(TOP*144.0)/(SUM**(.5))
C
C    Now solve for the heat transfer coefficient
C
        AITCH=QO/(TONE-TAW)
C
        WRITE(*,10) QO
        WRITE(*,11) AITCH
10      FORMAT(//' QO = ',1E12.7,' Btu/ft-squared-second')
11      FORMAT(' h = ',1E14.7,' BTU/FT-SQUARED-SEC-degR')
        STOP
5       FORMAT(1X,6(F8.5,3X))
        END
```

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Vita

Gerald W. Wirsig [redacted] in 1970 [redacted] served in the U.S. Air Force until 1976. He then attended the University of Kansas and was graduated with distinction with a Bachelors of Music Education degree in 1980. He attended Air Force Officer Training School, where he was the First Honor Graduate, and re-entered active duty in August, 1982. His first assignment was the Undergraduate Conversion Program at the Air Force Institute of Technology (AFIT), where he was a Distinguished Graduate, earning a Bachelors degree in Astronautical Engineering in March, 1984. Subsequently he served, first as a systems engineer, and then as chief of the launch accessories branch for the 6595th Space Shuttle Test Group at Vandenberg AFB, California until March, 1987. He was then selected to study Astronautical Engineering in the Masters program at AFIT.

[redacted]

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4. PERFORMING ORGANIZATION REPORT NUMBER(S) AFIT/GA/AA/88D-14	5. MONITORING ORGANIZATION REPORT NUMBER(S)
6a. NAME OF PERFORMING ORGANIZATION School of Engineering 6b. OFFICE SYMBOL (If applicable) AFIT/ENY	7a. NAME OF MONITORING ORGANIZATION
6c. ADDRESS (City, State, and ZIP Code) Air Force Institute of Technology Wright-Patterson AFB, OH 45433	7b. ADDRESS (City, State, and ZIP Code)
8a. NAME OF FUNDING/SPONSORING ORGANIZATION 8b. OFFICE SYMBOL (If applicable)	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER
8c. ADDRESS (City, State, and ZIP Code)	10. SOURCE OF FUNDING NUMBERS PROGRAM ELEMENT NO. / PROJECT NO. / TASK NO / WORK UNIT ACCESSION NO.

11. TITLE (Include Security Classification)
AN EXPERIMENTAL METHOD OF TESTING THIN-FILM HEAT TRANSFER GAGES (UNCLASSIFIED)

12. PERSONAL AUTHOR(S)
Wirsig, Gerald Wayne

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
MS Thesis	FROM ____ TO ____	1988 December	65

16. SUPPLEMENTARY NOTATION

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Heat Transfer, Thin-film Gages, Preheating, Two-Dimensional Effects
20	13		

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

Thin-film gages have been used successfully for measuring rapid temperature fluctuations in experiments with a time duration of a few milliseconds. For these short times, a one-dimensional, semi-infinite solid analysis may be used to convert surface temperatures into surface heat fluxes. However, two primary factors can cause inaccurate results: (1) gage preheating due to electrical currents in the gage, and (2) extended test times which allow two-dimensional effects to invalidate the use of the one-dimensional model. The objectives of this thesis are the determination of test time and electrical current limitations of thin-film heat transfer gages.

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